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Energetic and Exergetic Assessment of Biomass Integrated Gasification Combined Power and Ejector – Absorption Refrigeration System

Mohd Parvez

Email: mparvezalig@rediffmail.com
Department of Mechanical Engineering, Al-Falah University, Faridabad (Haryana) India

Abstract – In the present study, a new thermodynamic cycle for effective exploitation of biomass gasification potential is explored which combines the combined power cycle with an ejector, and an absorption refrigeration cycle in order to increase the overall energy conversion efficiency. This exergy based performance analysis is conducted to investigate the effects of overall pressure ratio, turbine inlet temperature in the overall cogeneration cycle to identify the causes and locations of thermodynamic imperfection. The results obtained from the analysis show that both energetic and exergetic efficiencies of the cogeneration cycle significantly vary with the change in gas turbine inlet temperature and decreases with the change in steam turbine inlet pressure but the change in biomass material shows small variation in these parameters.

Keywords: biomass, ejector, absorption refrigeration, gasification, solid waste, rice husk
Corresponding author:

1. INTRODUCTION

In recent years, it has been observed that there is a simultaneous demand for electric power and cooling for human comfort and his activities all over the world at the domestic and industrial purposes. To maintain this necessary power and cooling demand, renewable energy system is often arranged in combination with fossil fuels, which include coal, petroleum and natural gas, are the main resources to supply the energy. Although there is side effects of using fossil fuels have been seen such as global warming and climate change are clearly associated with their applications in power industry (Baratieri et al. 2008). Among all the renewable sources of energy, biomass has great potential as a sustainable energy for producing electricity and produce very low levels of particulates, NOx and Sox compared to the fossil fuels (Paisley et al. 2003). In the present study, biomass integrated gasification combined cycle has been used for production of electric power and to achieve the higher efficiency.

Gasification of biomass fuels has increased dramatically in recent years, and growth will continue for many decades as raising global energy demand, economic pressures and environmental legislation encourages the use of energy efficient, environmentally sound technology for production of heat and power production. The use of biomass gasification process is a key element in an advanced gas turbine combined cycle system (Mark and Mike 2003). Anil et al. (2006) solved the equations containing four atom balances (C, O, H and N) and equilibrium relations for gas compositions using MATLAB at atmospheric condition. Jarungthammachote and Dutta (2007) developed a thermodynamic equilibrium model based on equilibrium constants for predicting the composition of synthetic gas in a downdraft waste gasifier. Odukoya, et al. (2011) analyzed and shows the benefit of solid oxide fuel cell integration to IGCC plant. Parvez and Tasmeem (2013) study a thermo – chemical model based on equilibrium constants for the gasification of different biomass materials to find out the composition of syngas. Recently, Srinivas et al. (2015) carried out, the thermal performance of a biomass plant and find out that the supplementary firing in a combined cycle power plants results in a gain in power but in a loss in efficiency. In their study it has been found that lower values of RAFR, SFR, and compressor pressure ratio results favorable conditions to the IGCC plant.

In all these cycles it is observed that a significant amount of heat is wasted to the environment at the exit of heat recovery vapour generator of the cogeneration cycle and these exhaust losses destroy the work producing potential of combined power cycle which results in its inefficiency. It is pertinent to recover this waste heat which would otherwise be wasted. The current research explores the application of waste heat-driven absorption cooling system in the organic Rankine cycle with ejector. It is expected that the deployment of absorption cooling system at the bottom of combined ORC – Ejector cycle will reduce the energy consumption, increase product recovery, and improve overall energy conversion efficiency.

In view of the above, in the present study a cogeneration cycle is proposed for the simultaneous that produces both refrigeration and power in various range temperatures. The entire cycle is single source waste heat driven and produced refrigeration through ejector in the range – 1 ^{0}C to – 9 ^{0}C and air conditioning through single effect vapour absorption in the range 5 ^{0}C to 15 ^{0}C. Various thermodynamic investigations have been made in the few years on combined power and cooling cycle which integrate the ejector refrigeration system and vapour absorption refrigeration system (VARS) which is more advantageous compared to VARS due to higher initial cost of VARS (Eames et al.1995; Sankarlal and Mani, 2007; Parvez and Khaliq, 2014).

To the best of the author's knowledge an extensive research is reported in the literature for the development and analysis of combined power and ejector – absorption refrigeration cycles for the power generation as well as utilization of waste heat effectively. Hasan et al. (2002) investigated the first and second law of thermodynamics to optimize the combined cycle. A characteristic of this cycle is that the ammonia – water vapour that leaves the turbine passes through a heat exchanger transferring sensible heat, therefore, the produced cooling is relatively small. In order to produce a larger cooling effect, the working fluid should go through a phase change in the cooler. Sharifi and Khalilarya (2016) presented a novel combined heating, power and absorption – ejector refrigeration cycle driven by biomass fuel. The results show that the turbine inlet ammonia – water concentration, turbine outlet mass flow rate and turbine efficiency have effects on the turbine power output, refrigeration output, efficiency and exergy destruction in each component in the combined cycle.

Recently, an energy and exergy analyses of combined power and ejector – absorption refrigeration cycles was reported by (Berhane et al., 2010; Khalid et al., 2015; Sun et al. 2017).

The main objective of present study is to obtain the energy and exergy analysis of biomass gasification combined power and ejector – absorption refrigeration cycle to obtain further improvement in the efficiency of combined power and ejector – absorption refrigeration cycle.

The proposed combined ORC – ejector – absorption cycle is one of the great solutions of meeting out the increasing energy demand of power generation and air-conditioning through the effective exploitation of biomass integrated gasification power plant potential of the world. Most of the industries and buildings require power generation and cooling simultaneously like, oil and gas industry, petrochemical facilities, hotel resorts, hospitals etc. ORC turbine produce the power which may be delivered to the grid and ejector – absorption cycle provide cooling for air – conditioning of buildings through the utilization of biomass integrated gasification thermal energy as a primary energy input and the use of the eco-friendly refrigerants (R 141b) in a highly sustainable and environment friendly manner. Computational analysis was performed to investigate the effects of gas turbine inlet temperature, steam turbine inlet pressure, and change in biomass material on energetic and exergetic efficiencies of biomass fuelled ejector – absorption cycle are graphed and commented upon.

2. SYSTEM DESCRIPTION

The proposed cycle integrates the combined power R141b operated ejector refrigeration system and single effect absorption refrigeration cycle is shown in Fig.1. The biomass is injected to the gasifier at ambient conditions. The biomass gasification occurs in the presence of compressed air at 2 and superheated steam at 4, produces the syngas and goes to combustion chamber at 5 after passing through a gas clean up unit. The syngas burned in the combustion chamber in the presence of compressed air, and the combustion products at 6 goes to gas turbine where they expand and produce power. The gas turbine exhaust at 7 enters the HRSG where steam is generated. The superheat steam at 'a', goes to steam turbine for additional power production. Saturated steam at

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the exit of steam turbine at 8 goes to the ejector for power production and after routed goes to condenser where its phase changes from vapour to liquid. The water is then pumped to HRSG. The hot gases coming out from HRSG at state 15 fed to the generator of the vapour absorption system. The refrigerant (H_2O) is separated from LiBr–H_2O in the generator by means of the heat driven by the hot gas. After the refrigerant has reached the desired temperature it goes through the condenser 2 at 17 and evaporator 2 at 19 through the expansion valve at 18. The saturated steam at 20 enters the absorber where it mixes with a weak solution at 26, generating heat that has to be dissipated to increase the efficiency of the mixing process. The mixing process results in a strong solution that exits the absorber at 21 and is pumped to the upper pressure of the cycle at 22. The high pressure strong solution at 22 is heated to a high temperature. Stack gases at the exit of the generator discharge to the ambient at state 16.

For the analysis of the cycle the following assumptions have been made

1. Air fuel ratio in the gasifier is assumed to be equal to 0.5
2. Velocities of streams at the inlet and outlet of the ejector could be negligible
3. Mixing process in the mixing chamber of ejector occurs at constant pressure and complies with the conservation of energy and momentum
4. Lithium bromide solutions in the generator and in the desorber are assumed to be in equilibrium at their respective temperatures and pressures
5. Refrigerant at the condenser and the evaporator exits in saturated states
6. Strong solution of the refrigerant leaving the absorber and the weak solution of refrigerant leaving the generator are saturated.
7. To avoid crystallization of the solution, the temperature of the solution entering the throttle valve should be the least above crystallization temperature (7–8 ^{0}C)
8. The system uses stack heat at state 15 to drive the generator, which produces chilled water in the evaporator
9. Enthalpy values at various state points of the ejector cycle for a given refrigerant (R-141b) are taken from REFPROP 6.01(1998)

3. ENERGETIC AND EXERGETIC ANALYSIS OF COMBINED POWER CYCLE

Absolute entropy of the species 'i' and a mixture at a given (T, p) is calculated by the relations (Borgnakke and Sonntag 2009)

$$\bar{s}_i(T,p) = \bar{s}_i^0(T_0,p_0) + \int_{T_0}^{T} \frac{\bar{C}_{p_i}(T)}{T} dT - \bar{R} ln\left(\frac{y_i p}{p_0}\right) \quad (1)$$

$$\bar{s}_{mix}(T,p) = \sum_{i=1}^{i=n} y_i \bar{s}_i(T,p) \quad (2)$$

$$\frac{\bar{C}_{p_i}(T)}{\bar{R}} = \alpha + \beta T + \gamma T^2 + \delta T^3 + \varepsilon T^4 \quad (3)$$

$$\bar{C}_{p,mix} = \sum_{i=1}^{i=n} y_i \, \bar{C}_{p_i} \quad (4)$$

Compressor delivery pressure p_2 is 12 bar and temperature of air at the exit of compressor is

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_c \gamma}} \quad (5)$$

Power input in the compressor of admitted air is

$$\dot{W}_C = \dot{m}_{air}\left[\alpha_{air}(T_2 - T_1) + \frac{\beta_{air}}{2}(T_2^2 - T_1^2) + \frac{\gamma_{air}}{3}(T_2^3 - T_1^3) + \frac{\delta_{air}}{4}(T_2^4 - T_1^4)\right] \quad (6)$$

Chemical formula for biomass feedstock is given by $C_{a0}H_{a1}O_{a2}N_{a3}$, for single–carbon–atom fuel (a_0=1). The chemical reaction in gasifier is

$$C_{a0}H_{a1}O_{a2}N_{a3} + wH_2O + a_4(O_2 + 3.76N_2) + a_5H_2O \Rightarrow b_1CH_4 + b_2CO + b_3CO_2 + b_4H_2 + b_5H_2O + b_6N_2 \quad (7)$$

Complete combustion for syngas in gas turbine combustion chamber is

$$b_1CH_4 + b_2CO + b_3CO_2 + b_4H_2 + b_5H_2O + b_6N_2 + a_7(O_2 + 3.76N_2) \rightarrow b_7CO_2 + b_8H_2O + b_9N_2 + O_2 \quad (8)$$

Calorific value of syngas at a given combustion outlet temperature is

$$\sum_R \left(\bar{h}_f^0 + \Delta\bar{h}\right) + \bar{Q}_{cv} = \sum_P \left(\bar{h}_f^0 + \Delta\bar{h}\right) \quad (9)$$

Gas turbine outlet temperature is calculated after using the adiabatic expansion relation and pressure ratio across the turbine is

$$\dot{W}_{GT} = \dot{m}_{mix}\left[\alpha_{mix}(T_6 - T_7) + \frac{\beta_{mix}}{2}(T_6^2 - T_7^2) + \frac{\gamma_{mix}}{3}(T_6^3 - T_7^3) + \frac{\delta_{mix}}{4}(T_6^4 - T_7^4)\right] \quad (10)$$

Work produced by steam turbine is

$$\dot{W}_{ST} = \eta_{ST,is}(\dot{m}_a h_a - \dot{m}_8 h_8 - \dot{m}_c h_c) \quad (11)$$

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3.1 ENERGETIC AND EXERGETIC ANALYSIS OF EJECTOR AT THE EXIT OF STEAM TURBINE

Entrainment ratio based on mass, momentum and energy equation is reported by Dai et al. (2009)

$$\mu = \sqrt{\eta_n \eta_{\dot{m}} \eta_d \left(h_{pf,n1} - h_{pf,n2,\acute{s}}\right)/\left(h_{\dot{m}f,d,\acute{s}} - h_{\dot{m}f,\dot{m}}\right)} - 1 \quad (12)$$

Energy conservation equation for the adiabatic and steady primary flow is

$$\dot{m}_{pf} h_{pf,n2} + \frac{\dot{m}_{pf} u_{pf,n2}^2}{2} = \dot{m}_{pf} h_{pf,n1} + \frac{\dot{m}_{pf} u_{pf,n1}^2}{2} \quad (13)$$

Nozzle efficiency is

$$\eta_n = \frac{h_{pf,n1} - h_{pf,n2}}{h_{pf,n1} - h_{pf,n2,\acute{s}}} \quad (14)$$

Momentum conservation equation is

$$\dot{m}_{pf} u_{pf,n2} + \dot{m}_{sf} u_{sf,n2} = \left(\dot{m}_{pf} + \dot{m}_{sf}\right) u_{mf,m,\acute{s}} \quad (15)$$

Diffuser energy equation is

$$\frac{1}{2}\left(u_{mf,m}^2 - u_{mf,d,\acute{s}}^2\right) = h_{mf,d,\acute{s}} - h_{mf,m} \quad (16)$$

Diffuser efficiency is

$$\eta_d = \frac{h_{mf,d,\acute{s}} - h_{mf,m}}{h_{mf,d} - h_{mf,m}} \quad (17)$$

Work consumed by the feed pump to increase its pressure from 10 to b is

$$\dot{W}_{FP1} = \dot{m}_c (h_b - h_{10}) \quad (18)$$

3.2 ENERGETIC EXERGETIC ANALYSIS VAPOUR AND ABSORPTION REFRIGERATION AT THE EXIT OF HEAT RECOVERY STEAM GENERATOR

Energy and mass balances to determine the heat transferred by the external fluid to the solution within the generator is

$$\dot{m}_{s15} h_{23} + \dot{m}_G h_{g15} - (\dot{m}_{s15} - \dot{m}_r) h_{24} - \dot{m}_r h_{17} - \dot{m}_G h_{g,leaving\ GEN} = 0 \quad (19)$$

Exergy of the fluids that enters the generator is

$$\sum \dot{m}_{in,G} e_{in,G} = \dot{m}_G [(h_{15} - h_0) - T_0(s_{15} - s_0)] + \dot{m}_{s15} [(h_{23} - h_0) - T_0(s_{23} - s_0)] \quad (20)$$

Exergy of the outgoing mass flow is

$$\sum \dot{m}_{out,G} e_{out,G} = (\dot{m}_{s15} - \dot{m}_r)[(h_{24} - h_0) - T_0(s_{24} - s_0)] + \dot{m}_G\left[\left(h_{g,leaving\ GEN} - h_0\right) - T_0\left(\dot{s}_{g,leaving\ GEN} - s_0\right)\right] + \dot{m}_r [(h_{17} - h_0) - T_0(s_{17} - s_0)] \quad (21)$$

3.3 PERFORMANCE PARAMETERS OF VARIOUS COMPONENTS

The energetic efficiency of combined power and ejector – absorption refrigeration cycle is

$$\eta = \frac{\dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{AC} - \dot{W}_{FP1} - \dot{W}_{P2} + \dot{Q}_{E1} + \dot{Q}_{E2}}{\dot{m}_{fuel} LHV} \quad (22)$$

where $\dot{m}_{fuel}$ is the mass flow rate of fuel consumed, $\dot{Q}_{E1}$ cooling of the ejector evaporator and $\dot{Q}_{E2}$ is the amount of cooling produced at the evaporator of vapour absorption refrigeration

The exergetic efficiency of combined power and ejector – absorption refrigeration cycle is

$$\Psi = \frac{\dot{W}_{GT} + \dot{W}_{ST} + \dot{W}_{RT} - \dot{W}_{AC} - \dot{W}_{FP1} - \dot{W}_{P2} + \dot{E}_E + \dot{E}_R}{\dot{E}_{fuel,in}} \quad (23)$$

where $\dot{E}_{fuel,in}$ is the exergy of fuel, $\dot{E}_E$ is the amount of exergy associated with the refrigeration capacity ($\dot{Q}_E$) of the ejector evaporator and $\dot{E}_R$ is the exergy of refrigeration which is the refrigeration capacity $\dot{Q}_E$ and T_{EV} is temperature of evaporator

The efficiencies of nozzle, mixing chamber and diffuser are reported in Table 1. The composition of biomass materials is reported in Table 2 and coefficient of syngas gas is given in Table 3.

4. RESULTS AND DISCUSSION

For the numerical appreciation of the results, in this study, a parametric analysis is performed to assess the effect of gas turbine inlet temperature, steam turbine inlet pressure, and change in biomass material on the energetic efficiency of biomass gasification based combined power and ejector – absorption refrigeration cycle. Energetic efficiency and energy distribution of biomass fuel is obtained by energy balance analysis of the cycle.

Figs. 2-3 shows the variation of energetic and exergetic efficiencies of biomass gasification triple power cycle for the cases of with and without considering the ejector – absorption refrigeration system with the change in gas turbine inlet temperature. In general, exergetic efficiencies of both triple power and combined power and ejector – absorption refrigeration cycles are slightly lower than their energetic efficiencies. This is due to the fact that the chemical exergy of fuel (biomass) which is considered as the input in exergy analysis is slightly higher than the calorific value of the fuel which is considered as the input in the energy analysis. On comparing the performance of triple power cycle with and without ejector – absorption refrigeration system, energetic efficiency has improved considerably while exergetic efficiency of the triple power cycle increased marginally with the integration of ejector – absorption refrigeration system. This is due to the reason that amount of exergy associated with the refrigeration capacity of the ejector – absorption is considerably lower than the energy of the refrigeration capacity. In general, it is observed that employment of ejector – absorption in triple power cycle on an average enhances its energetic efficiency by 6.8 % for both the biomass materials considered in the analysis. On the other hand exergetic efficiency of the same cycle on an average increased by 4.4 % due to the reasons explained above. Figs. 2-3 reveal that both energetic and exergetic efficiencies increase linearly as gas turbine inlet temperature increases for both the cases of biomass considered. It is further noticed that both energetic and exergetic efficiencies of triple power cycle and combined power and ejector – absorption cooling cycle are marginally higher for solid waste and lower for rice husk. This is due to the fact that gasifier temperature and LHV of syngas is higher in case of solid waste and considerably lower in case of rice husk.

Figs. 4-5 shows the variation of energetic and exergetic efficiencies of triple power cycle with the change in steam turbine inlet pressure. It is found that both energetic and exergetic efficiencies decreases with the increase in steam turbine inlet pressure. The reason for this kind of trend is that increase in turbine inlet pressure results in lower mass flow rate of steam produced in the heat recovery steam generator which in turn reduces the steam turbine output and hence decreases the overall efficiency of the cycle. Since the contribution of gas turbine towards the overall power generation is much higher and is three times larger than the contribution of the steam turbine, and change in steam turbine pressure only effects the steam turbine output not the gas turbine output, therefore, energetic efficiency of triple power cycle slightly drops with the increase in steam turbine inlet pressure. For the similar reasons, exergetic efficiency also drops slightly with the same. An effect of ejector – absorption employment in triple power cycle for different steam turbine inlet pressures on both energetic and exergetic efficiencies is also shown in the above figures. It is shown that energetic efficiency of triple power cycle for solid waste was obtained as 38.73%, alternatively it increases to 45.79% after the employment of ejector – absorption refrigeration. For rice husk fuelled triple power cycle the efficiency increases from 35.66% to 42.75% after the ejector – absorption employment. It is also observed that exergetic efficiency of triple power cycle decreases from 41.73 % to 35.48 % when biomass is changed from solid waste to rice husk. This is due to the reason that chemical exergy of biomass reduces as moisture content increased as shown in Table 1.

4.1 CONCLUSIONS

Energy and exergy analysis of a biomass gasification fuelled triple power thermodynamic cycle using the superimposition of IGCC and ejector – absorption refrigeration cycle are conducted. The performance of the system is examined under the variation of gas turbine inlet temperature and steam turbine inlet pressure. The main concluding remarks from this study are as follows:

- Both energetic and exergetic efficiencies of triple power cycle increases linearly with the increase in gas turbine inlet temperature.
- There is on average, 6.8% gain in energetic efficiency and 4.4% gain in exergetic efficiency of triple power cycle when ejector – absorption refrigeration cycle is employed.
- The performance of triple power cycle is insensitive to the variation of the biomass as a primary fuel input. Slight variation in both energetic and exergetic efficiencies were obtained when the biomass is changed from solid waste to rice husk.
- Both energetic and exergetic efficiencies of triple power cycle slightly decreases when steam turbine inlet pressure increased from 30 bar to 50 bar.
- Energy and exergy distribution more or less shows the same trends for all considered biomass materials in a proposed triple power and cooling cycle.

NOMENCLATURE

AFR   Air-fuel ratio

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C Compressor, Carbon
C1 Steam condenser
C2 Refrigerant condenser
CC Combustion Chamber
$\dot{E}$ Exergy rate (kW)
E Specific exergy (kJ/kg)
GT Gas turbine
HRSG Heat recovery steam generator
HRVG Heat recovery vapor generator
h Specific enthalpy (kJ/kg)
$\Delta\bar{h}$ Change in enthalpy of a species or mixture from ambient state to given state (kJ/k mol)
$\bar{h}_f^0$ Enthalpy of formation of species of a mixture at ambient conditions (kJ/k mol)
IGCC Integrated gasification combined cycle
$\dot{m}$ Mass flow rate (kg/s)
$\dot{m}_{vapor}$ Mass flow rate of refrigerant (R–141b) vapor in ejector refrigerant cycle (kg/s)
ORC Organic Rankine cycle
P Product
P1 Pump1
P2 Pump 2
$\dot{Q}$ Heat transfer rate (kW)
$\bar{Q}_{cv}$ Calorific value (kJ/kmol)
$\dot{Q}_E$ Refrigeration effect in refrigerator (kW)
R Reactant
RT Refrigerant turbine
ST Steam turbine
s Specific entropy (kJ/kg K)
$\dot{S}_{gen}$ Entropy generation rate (kW/K)
T Absolute temperature (K)
TV Throttling valve
T_p Saturated temperature at pressure of process steam (K)
u Velocity(m/s)
$\dot{W}$ Power (kW)
η_I Energetic efficiency
η_{II} Exergetic efficiency
μ Entrainment ratio in the ejector

Suffix

a–e State points of the steam cycle
bmf Biomass fuel
C Compressor
Ch Chemical
d.a.f. Dry ash free
E Evaporator
F Formation, fuel
d Diffuser
n Nozzle
m Mixing chamber
pf Primary flow
sf Secondary flow
n1 Inlet of nozzle
n2 Outlet of nozzle
s' Isentropic
Φ Exergy ratio
$\alpha, \beta, \gamma, \delta, \varepsilon$ Constants
0 Reference point
1–18 State points of the triple power and cooling cycle
$(-)$ Per mol

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Table1. Thermo-chemical properties of biomass materials

Property	Units	Solid Waste	Rice Husk
C	Wt% daf	51.03	36.42
H	Wt% daf	6.77	4.91
O	Wt% daf	39.17	35.88
N	Wt% daf	2.64	0.59
Lower Heating Value (LHV)	kJ/kg,daf	25021.51	22982.32
Moisture Content	Wt%	16.00	12.00
Ash	Wt%	5.00	22.20
Exergy Ratio		1.037	1.133

Table2. Properties and operating variables for the analysis of the proposed power and cooling cycle configuration

Equipment Performance	
Isentropic efficiency of compressor ($\eta_{c,isen}$)	85%
Isentropic efficiency of gas turbine ($\eta_{gt,isen}$)	85%
Isentropic efficiency of steam turbine ($\eta_{st,isen}$)	85%
Pressure drop across the gasifier (ΔP_G)	5%
Pressure drop across the combustion chamber (ΔP_{CC})	3%
Pressure drop across the HRSG (ΔP_{HRSG})	2%
Steam cycle pump isentropic efficiency	85%
Operating Parameters	
Compressor pressure ratio (r_p) bar	12 (fixed)
Turbine inlet temperature TIT (^{0}C)	1000–1200 (range)

Condenser pressure (bar)	0.06
Pinch point temperature	30 K (fixed)
Approach temperature	15^0C
Steam turbine inlet pressure (bar)	30 – 70
Refrigerant turbine isentropic efficiency	80%
ORC pump isentropic efficiency	85%
Nozzle efficiency	90 %
Mixing chamber efficiency	85 %
Diffuser efficiency	85 %

Table3. Composition of syngas produced after gasification and exhaust gas after composition for one kmol of biomass at pressure ratio (r_p=12)

Constituent	Solid Waste Concentration (kmol)	Rice Husk Concentration (kmol)
Synthetic Gas		
CH_4	b1=0.16	b1=0.06
CO	b2=0.52	b2=0.62
CO_2	b3=0.32	b3=0.31
H_2	b4=0.84	b4=0.99
H_2O	b5=0.94	b5=1.12
N_2	b6=0.44	b6=0.39
Exhaust Gas (Combustion Product)		
CO_2	b7=1	b7=1
H_2O	b8=2.1	b8=2.23
N_2	b9=6.08	b9=5.65
O_2	0.5	0.47
a_7	1.5	1.4

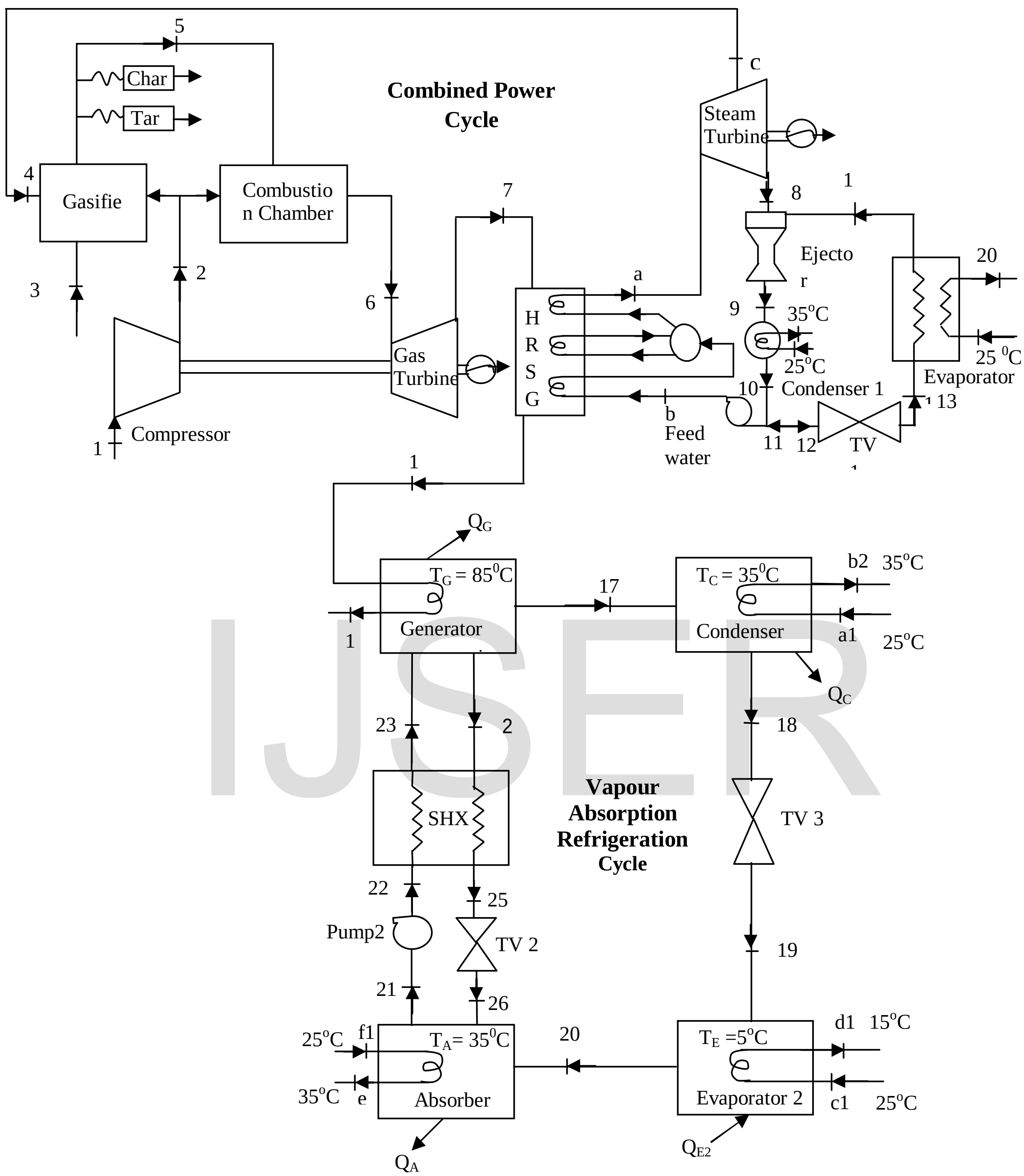

Fig. 1 Combined power and ejector – absorption refrigeration cycle

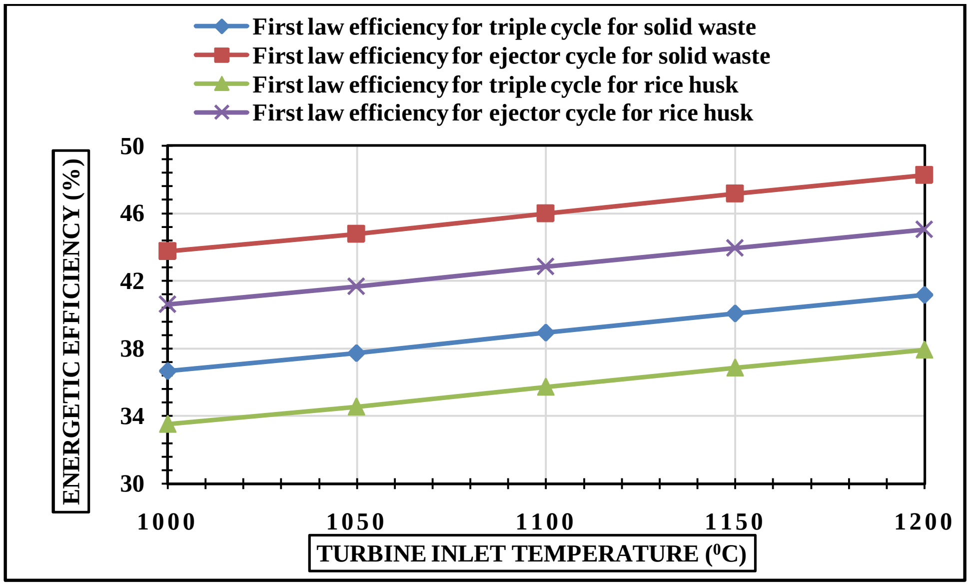

Fig. 2 Variation of energetic efficiency of combined power and ejector – absorption refrigeration cycle with turbine inlet temperature at (r_p=12)

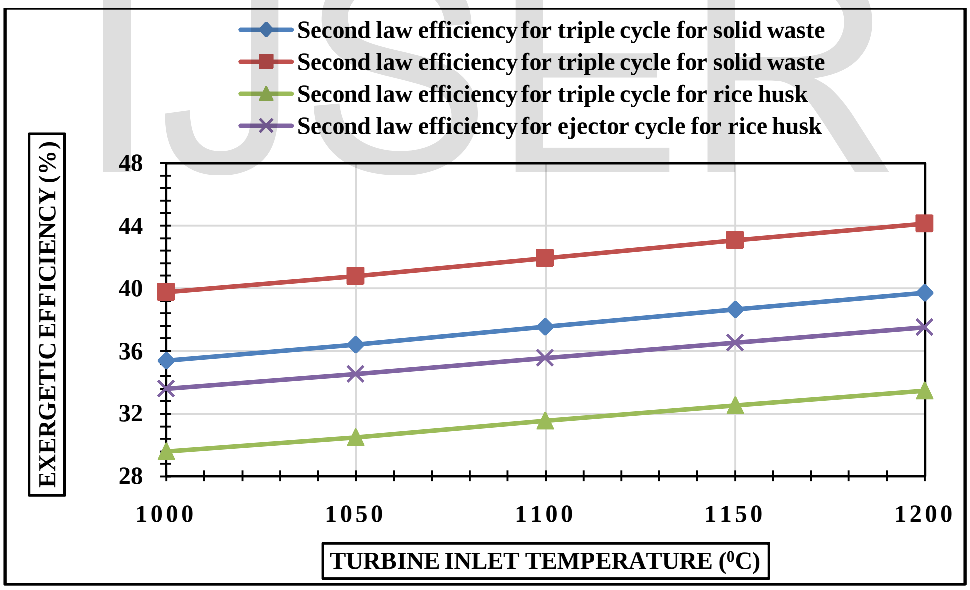

Fig. 3 Variation of exergetic efficiency of combined power and ejector – absorption refrigeration cycle with turbine inlet temperature at (r_p=12)

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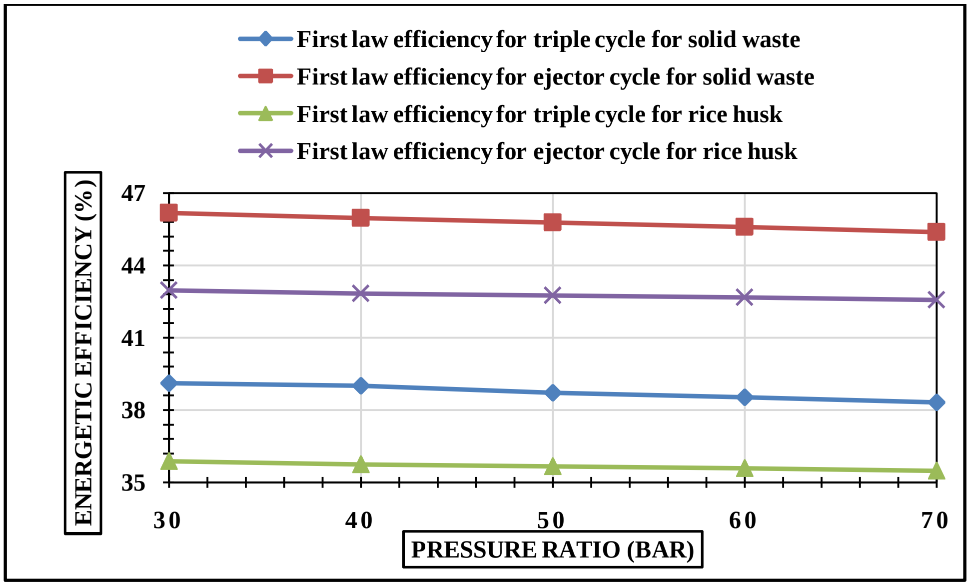

Fig. 4 Variation of energetic efficiency of combined power and ejector – absorption refrigeration cycle with steam turbine inlet pressure at (TIT=1100 ^{0}C)

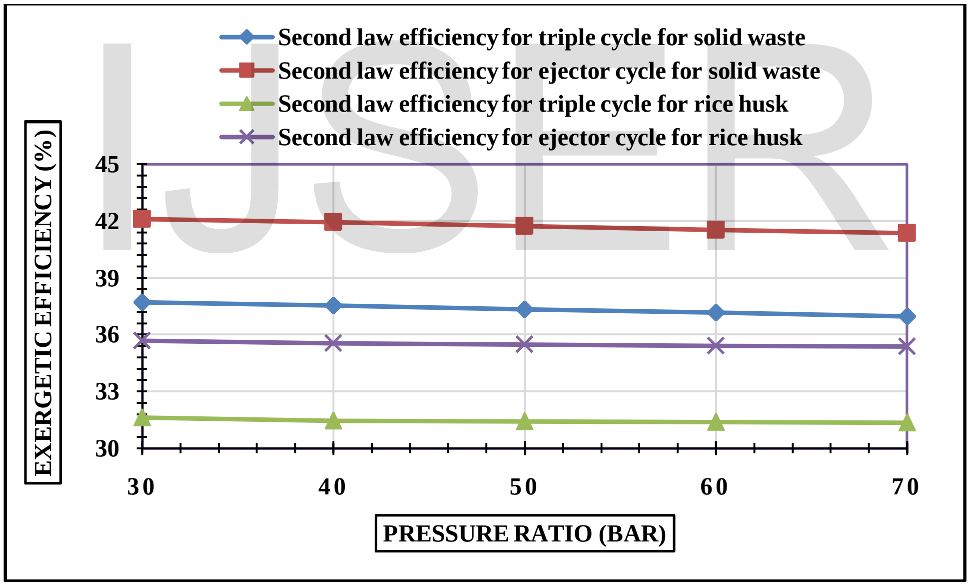

Fig. 5 Variation of exergetic efficiency of combined power and ejector – absorption refrigeration cycle with steam turbine inlet pressure at (TIT=1100 ^{0}C)